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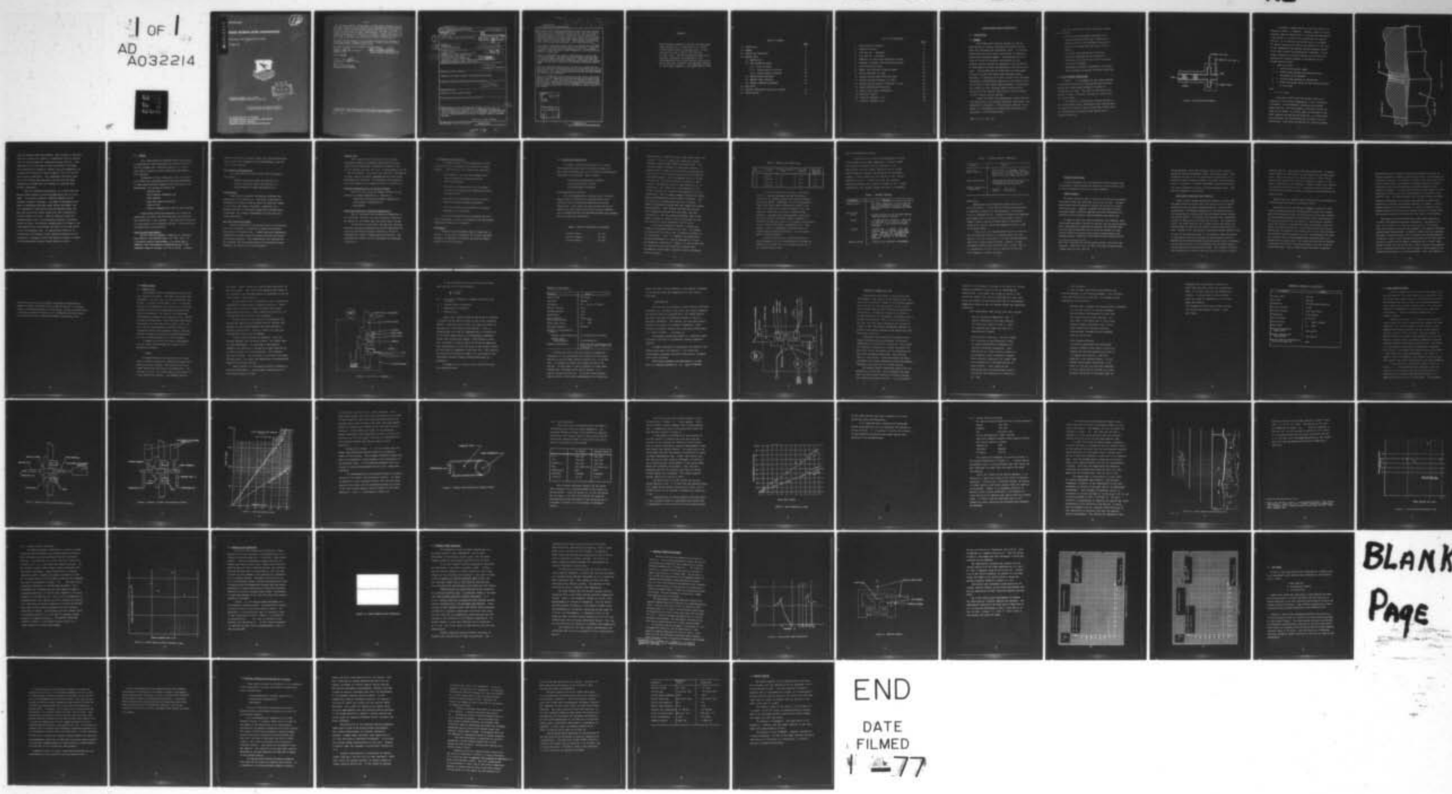

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LIQUID BEARING GYRO INVESTIGATION

THE SINGER COMPANY, KEARFOTT DIVISION

OCTOBER 1976

AIR FORCE AVIONICS LABORATORY

TECHNICAL REPORT AFAL-TR-75-203
REPORT FOR PERIOD APRIL 1974 - JUNE 1975

Approved for public release; distribution unlimited

AIR FORCE AVIONICS LABORATORY
AIR FORCE WRIGHT AERONAUTICAL LABORATORIES
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20. ABSTRACT (Continue on reverse side if necessary and identify by block number)

THE OBJECTIVE OF THIS PROGRAM WAS TO FABRICATE AND TEST TWO LIQUID HYDRODYNAMIC BEARINGS TO BE SUBSEQUENTLY INCORPORATED INTO A LIQUID BEARING GYROFLEX* GYRO TO BE EVALUATED WITH THE POSSIBILITY OF FUTURE RETROFIT INTO THE KT-70 SERIES INERTIAL GUIDANCE SYSTEMS.

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The objective of this program was to fabricate and test two liquid hydrodynamic bearings to be subsequently incorporated into a liquid bearing gyro.

The liquid bearing design consists of a spiral grooved journal bearing with two "in-pumping" thrust plates. The low viscosity hydrocarbon bearing fluid is captured in the bearing gap by preloaded mercury ring seals at the ends of the journal shafts. The seal capability has been shown to be greater than one atmosphere under static conditions and greater than two atmospheres under dynamic, running conditions. Instrumented tests show the inherently, noise free operation of the liquid bearing.

Life tests of 840 hours and 400 hours on unloaded bearings were successfully performed without bearing degradation. Tests were performed to monitor bearing temperature rise, running power stability, self vibration, power consumption, and fluid and seal pressure fluctuations.

Tests were performed to determine bearing radial, axial, and angular stiffness. The bearing groove geometry was redesigned for improved load deflection capability. Tests confirmed the increased stiffness. Computer programs were devised to optimize design although no bearings of the optimized design were fabricated.

Vibration tests were performed on unloaded and loaded bearings with the vibration direction parallel to and perpendicular to the bearing spin axis. No resonances occurred which degraded bearing performance. No resonance was observed that was not identifiable as a test fixture resonance.

Testing of assembled liquid bearing gyros was usually terminated during initial calibration and dynamic balancing procedures due to seal failure. The interaction of the unbalanced rotor forces with the seal together with low angular stiffness disrupted the mercury seals. The design changes necessary to enhance the bearing angular stiffness could not be implemented before the end of the program.

PREFACE

This document reports on the work accomplished under Contract No. F33615-74-C-1136, titled "Liquid Bearing Gyro Investigation", in the time period from 17 April 1974 through 30 June 1975. The work performed under this contract represents an extension of the investigations into liquid bearing gyros which were internally funded by the Kearfott Divison of the Singer Company in the years 1968 to 1974.

TABLE OF CONTENTS

LIST OF ILLUSTRATIONS

LIQUID BEARING GYRO INVESTIGATION

1.0 INTRODUCTION

1.1 GENERAL

Field experience acquired through the various applications of inertial navigators throughout the Air Force has indicated high operational costs. It has been established that these costs are strongly influenced by recurring maintenance demands. The gyroscope has been found to be one of the major contributors to this problem; chiefly through an undesirably high incidence of catastrophic failures and degradation of accuracy with time. This problem can be alleviated with development of less noisy and more stable and repeatable rotor bearings. In recognition of this a project was undertaken to develop a hydro-dynamic liquid gyro rotor bearing that can be used in tuned rotor type gyroscopes. Specifically, this effort has been directed toward evaluation and evolution of the Kearfott Liquiflex for direct retrofit in KT-70 series inertial systems.

Under this effort two hydrodynamic liquid bearings were fabricated, with recently developed improvements, and subjected to evaluation. The evaluation consists of two parts; i.e. evaluation as a bearing and evaluation as a gyroscope (in the Gyroflex* gyro).

*Reg. in U. S. Pat. Off.

The basic objectives of this evaluation program are as follows:

- To verify the performance qualities of a liquid bearing Gyroflex gyro relative to a desired potential for navigator accuracy to within one nautical mile per hour (on a 3σ basis).
- To examine the physical and functional Liquiflex characteristics to assess its potential for retrofit into the KT-70 series of inertial systems.
- Recommend such design improvements or refinements required to achieve the above objectives.

1.2 LIQUID BEARING DESCRIPTION

Figure 1 is a schematic of the liquid bearing. It consists basically of a conventional spiral groove journal and thrust plate designed to operate on incompressible fluids. The fluid is retained in the bearing by means of mercury ring seals at each end of the bearing shaft.

The mercury is injected into annular grooves on the journal shaft with a sufficiently high preload head to prevent the volatile bearing fluid from leaking into the evacuated gyro enclosure under operational or storage conditions.

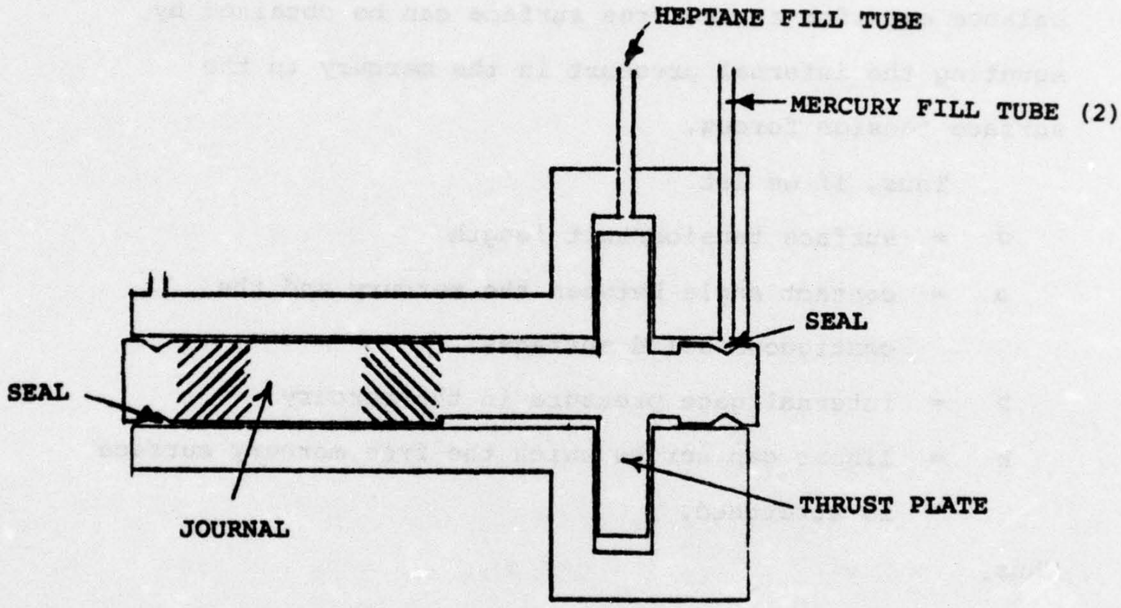

Figure 1 Liquid Bearing Schematic

A schematic representation of the seal area of the bearing is shown in **Figure 2**. Mercury, under an initial preload, forms an annular seal in a region defined between a Vee groove in the end of the journal and the bearing housing. If the pressure across the seal is increased, the mercury moves up one side of the groove with the area of the unsupported free mercury surface becoming smaller. If the free surface is considered as a membrane, a force balance equation at the free surface can be obtained by equating the internal pressure in the mercury to the surface tension forces.

Thus, if we let

σ = surface tension/unit length

α = contact angle between the mercury and the contiguous solid surfaces

P = internal gage pressure in the mercury

h = linear gap across which the free mercury surface is stretched.

Thus,

$$Ph = 2\,\sigma \cos\alpha$$

Since the contact angle and surface tension are invariant at a particular temperature, P will increase as h decreases. By continuously increasing the pressure across the seal, the free surface is forced to ascend the sides of the groove to a region where the decrease in gap width restores the equilibrium condition. At sufficiently high pressures, the free surface can be forced into the bearing gap. This condition defines the limiting pressure

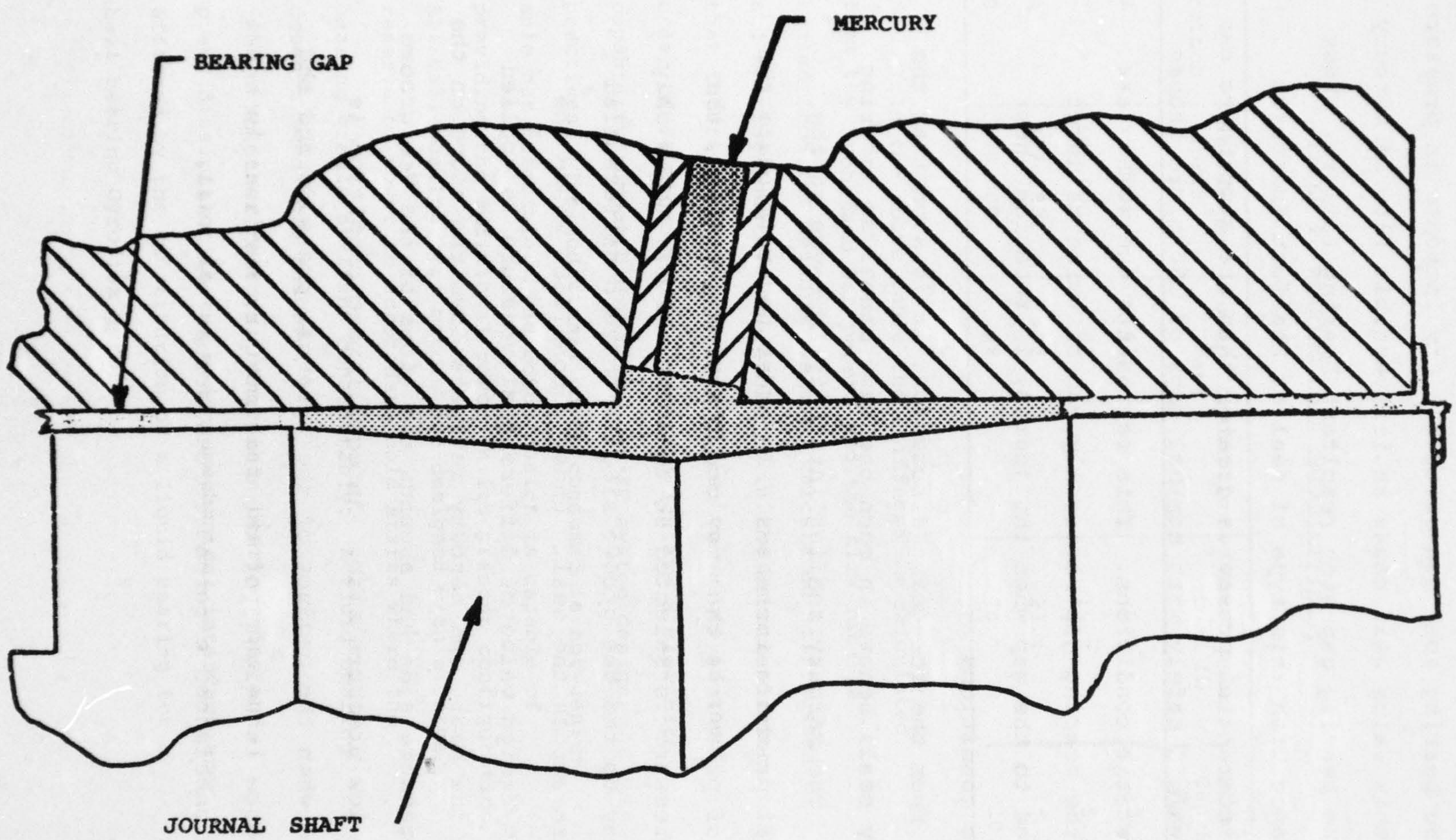

Figure 2 Section of Groove

that the bearing seal can sustain. Any increase in pressure above this value will cause an irreversible flow of mercury into the bearing gap with resultant bearing failure. Test experience with this type of seal at Kearfott indicates that differential pressures greater than one atmosphere can be sustained safely with nominal gaps of 150 micro inches under static conditions. This represents the worst case since the bearing gap statically can be twice as large compared to the gap when the journal is rotating under dynamic conditions.

From the foregoing discussion, it is seen that the mercury seals behave in much the same manner as "o" ring seals. The mercury sealing integrity improves as the internal pressure increases. In order to accommodate the range of pressures that may occur across the seals, the fill pressure is selected so that the mercury moves only part way up the Vee groove sides with zero differential pressure across the seal. Under a condition where the maximum design value of differential pressure is applied across the seal, the mercury retreats down the slope on the high pressure side and ascends the slope of the Vee groove on the low pressure side. An equilibrium condition is reached when the product of the internal pressure and the difference in heights of the free mercury surfaces is equal to the unbalanced external forces across the seal.

2.0 SUMMARY

This investigation is directed toward determining the applicability and practicality of retrofitting KT-70 inertial systems with "Liquiflex" gyros (i.e. a tuned rotor device containing liquid lubricated hydrodynamic spin bearings).

Fundamental to this investigation are the factors which affect gyro performance and life since degradation in these areas are prime causes of high maintenance cost. Specifically, the factors of concern are:

- Bearing noise
- Environmental susceptibility
- Load Capacity
- Spin axis torque variations
- Running life
- Physical compatibility with the KT-70 systems

These factors have been evaluated in a series of experiments and tests, the data from which are reported and interpreted in this document. The experiments conducted in relationship to the evaluation factors identified above are described in the following.

Bearing Noise Measurements

The gyro bearing assembly, installed in a Gyroflex gyro housing, instrumented along its spin axis by a vibration sensing accelerometer, is mounted upon a compliant base which permits unconfined motion in the frequency range of interest (i.e. 20 to 5K Hz). Internal

forces at work within the gyro cause rigid body accelerations which are directly measured by the accelerometer, (see Section 4.3 for results).

Environmental Susceptibility

The experiments and tests which serve to evaluate this factor

- Thermal sensitivity tests (see section 4.1)
- External Vibration tests (see section 4.5)
- Vacuum capability tests (see section 4.1)

Load Capacity

Load carrying capacity and bearing stiffness data arereported on in section 4.2. This data is obtained by pneumetically loading a bearing (while at rated spin speed) in both axial and radial directions. Deflection was measured by means of calibrated capacitive instrumentation so designed as to permit measurements in both axial and radial directions.

Spin Axis Torque Variations

Current variations in the driving coils of a hysteresis motor are, in general, indicative of torque fluctuations acting about the spindle axis. Current constancy measurement, therefore,is a handy means for quickly assessing the quality of gyro spin bearings. Such measurements were obtained for the Liquiflex and are recorded along with appropriate interpretation, in section 4.4 of this document.

Running Life

While bearing life tests can be very lengthy processes, properly instrumented and relatively short tests can give meaningful indications of bearing life. In this program two tests were run, of approximately 900 and 400 hrs duration. The latter test was fully instrumented in such a way as to detect impending failure long before performance was affected. No indications were observed. This test sequence is discussed in detail in section 4.1.

Physical Compatability with the KT-70 System

There are three areas of concern relevant to the question of system compatibility. These are:

- Mechanical/electrical interface compatibility
- Environmental susceptibility
- Performance

Mechanical/Electrical Interface Compatibility

This is concerned with the physical dimensions of the Liquiflex, its point to point electrical continuity, and impact of its demands upon the system power sources.

On the basis of analytical and empirical considerations,it has been determined that neither mechanical nor electrical interface variances relative to the KT-70 configuration need be imposed upon the Liquiflex. The considerations leading to this conclusion are discussed in section 6.

Environmental Susceptibility

Of primary concern is the compatibility of the Liquiflex with the environment within the KT-70 gimbal assembly. Specifically, this concerns the following:

- The composite vibration environment transmitted through the platform isolation system and gimbals.
- The operating temperature at the instrument cluster and the quality of temperature control provided therein.
- The heat flux load imposed by the Liquiflex upon the platform thermal environment.
- The "self vibration" presented to the gimbal assembly by the Liquiflex

The above considerations are discussed and their effects assessed in terms of anticipated life and performance prognostications in Section 6.

Performance

The basic gyro performance data is summarized in Section 5 along with a discussion of how each evaluated parameter is expected to influence the inertial memory quality of the KT-70 system.

3.0 RESULTS AND CONCLUSIONS

In order to establish the design of a liquid bearing Gyroflex gyro that can directly replace the ball bearing Gyroflex gyros in a KT series system,several areas must be considered. These include:

- Electrical and mechanical interface
- Bearing performance testing
- Gyro performance evaluation

Electromechanical Interface

The electromechanical interface study included a design study to establish the maximum bearing size consistent with the existing geometrical and power constraints. This effort has been carried out and the results are that a bearing with the following geometry can be retrofitted into the existing Gyroflex gyro package with a minimum cost and effort.

TABLE 1 BEARING GEOMETRICAL CONSTRAINTS

Journal length	-	.55 inch
Journal diameter	-	.25 inch
Thrust diameter	-	.48 inch

This bearing is somewhat shorter than those tested (17% shorter). This will impact both radial and angular stiffness reducing them 17%and 34% respectively. The effect on radial stiffness is considered tolerable. A reduction of 17% from the existing value established at 75000 lb/inch in the testing would yield a stiffness of over 62000 lb/inch. This corresponds to 47.5 g at 80% eccentricity. The loss of angular stiffness is more serious. The measured value of rate capability is 2.5 radians/sec. The effect of a 17% shortening of the journal by itself is to reduce this capability to 1.73 radians/sec. This is offset somewhat because the shorter bearing package permits larger angles to develop. The net effect is a maximum slew rate of 2.3 rad/sec. It makes the problem of system slew rate capability slightly more difficult. Older systems are slew rate limited at 25 radians per second while the newer ones are limited at 10. For any system considered, part of the effort involved in retrofit must be resetting the slew rate limit to a value compatible with the gyro.

It is not necessary to accept 2.3 radians per second as the maximum possible slew rate. For example a reduction of the journal gap to increase stiffness is an alternative. Table 2 compares the analytically derived performance of 4 journal bearings all fitting the geometric constraints in Table 1. It can be seen that by reducing the gap, and accepting increased power consumption, significantly improved journal performance can be achieved.

TABLE 2 BEARING LOAD CAPABILITIES

Gap Microinch	Stiffness lb/in	Power Consumption Watt	Max Linear Load Lb.	Slew Rate Capability Rad/sec
150	55,800	.41	6.7	2.3
100	188,000	.62	15.0	5.15
80	367,000	.77	23.4	8.1
70	540,000	.88	30.7	10.5

In the above table the stiffness is calculated from the Vohr and Pan paper, and the slew rate is scaled from the measured values, calculated stiffnesses and bearing length. Therefore, it is the recommendation that future journal gaps be modified to values in the order of 70 micro inch in order to achieve a slew rate capability compatible with recent KT system design practices. The additional power consumed is not desirable but tolerable. The motor design is capable of providing the additional power in its present configuration. All ball bearing gyro motors are designed with a large reserve of power to accomodate cold starts where the bearing grease stiffens considerably. A fluid bearing doesn't need this reserve.

No other factors, other than bearing size and power dissipation in the electromechanical interface are affected by the substitution of a liquid bearing for a ball bearing Gyroflex gyro.

Bearing Performance Testing

A large series of tests and measurements on the liquid bearing have been completed. In none of them are results indicated that are contrary to the basic thrust of this effort, namely to use a liquid bearing gyro to retrofit the KT series of platforms. Table 3 below summarizes the results of the testing completed to date. Please note that the journal bearing tested was over sized in length by 17%. Values adjusted for the correct length are given in brackets.

TABLE 3 BEARING TESTING

Parameter	Results
Bearing Life	Two tests completed, 30 days and 400 hrs of continuous running. Instrumentation showed no degradation after 400 hrs
Spring Rate	
Axial	27,500 lb/inch on non optimum design. Improved design available
Radial	75,000 lb/inch (62,000). This can be improved to as high as 540,000 lb/inch by a gap change and power increase
Angular	2.6×10^9 dyne cm/radian. This corresponds to 2.5 rad/sec slew rate (2.3). However, by a suitable gap change this can be increased to over 10 rad/sec in the shorter journal.
Bearing noise	1 milli g is routinely achievable

TABLE 3 BEARING TESTING (CONTINUED)

Parameter	Results
Spin Axis Torque Stability	Better than 0.1 milliamp current stability in a temperature controlled environment. Constant journal power demand to linear inputs up to 19 G.
Power Requirements	2 watts running measured (1.81 @ 150 μinch gap) (2.4 @ 60 μinch gap) 4.2 watts starting transient
Bearing Temperature Sensitivity	Mercury - .002 psi/°C Heptane - .001 psi/°C

Conclusions

These bearing performance data makes several conclusions possible. First, thrust bearing performance must be improved by adopting an optimum design. The present value is unacceptably low. Analysis indicates that adequate performance can be achieved with no increase in power. Second, journal bearing stiffness must be enhanced by reducing the journal gap to approximately 70 micro inches as discussed above.

Finally, all the data measured thus far as well as the criteria established in the design study, are compatible with the replacement of the ball bearing Gyroflex gyro with a liquid bearing version. However, no gyro performance measurements have been made to date. Thus it is recommended that any further judgments be held off until these data becomes available.

BEARING PROBLEM AREAS

A brief discussion will be presented on various problem areas encountered in bearing design and fabrication and the methods used to obtain acceptable solutions for these problems.

GROOVE GEOMETRY

The design equations used to obtain the desired bearing geometry were based largely upon published work by Vohr and Pan*. Suitable modifications were introduced to account for the use of a liquid rather than a gas. Computer, programs were developed to obtain optimized values of bearing compliance and power dissipation as a function of fluid viscosity, bearing geometry and bearing gap.

In order to carry over design information into hardware, techniques were developed to machine the grooves into the bearing surface with a high degree of precision and accuracy. The preferred technique for groove fabrication was to erode the grooves with high velocity argon ions through close fitting aluminum masks on the bearing surfaces. Apertures in the masks precisely duplicated the

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*Pan C.H.T. and Vohr J.H, "On the Spiral Grooved, Self Acting Gas Bearing", Report No. M.T.I. 63 TR52, (AD-433660) Prepared under Contact nonr 373 (00), Jan. 64.

desired geometry. With this technique, close control could be maintained over groove geometry, depth, and surface finish. With randomized scanning of the ion beam over the eroded areas, it is possible to obtain surface finishes of the order of one microinch R.M.S., if required. A further improvement in groove edge definition can be obtained by the use of photolithographic masks. This was not attempted on this program.

SEAL SURFACE FINISH AND SEAL INTEGRITY

Journal bearing designs developed by Kearfott initially used Vee shaped grooves at either end of the journal shift to capture pressurized rings of mercury thereby achieving a friction free rotating seal. Microscopic examination at low magnifications of the Vee grooves on ceramic coated journals showed surface pitting and surface defects. It was felt that during rotation microscopic globules of memory could be sheared off the seal. The sloping surface of the groove would act as a slinger forcing the sheared off globules away from the seal and thereby inhibiting healing of the seal. Because of the difficulties in machining annular grooves on the I.D bore of the bearing housing, it was decided to keep the grooves on the rotating shaft but to modify the groove shape so that exceptionally good surface finish and groove geometry could be obtained. The Vee grove geometry was not compatible with the random lapping motions required to achieve good finishes. This geometry, threfore, was characterized by surface asperities and/or microscopic circumf-

erential grooving. The groove sides were made arcuate to allow a greater degree of random motion in the lapping and polishing operations. With this modification, highly polished, surface finish grooves were obtained with finishes comparable in quality to the races in good quality ball bearings. Surface finishes were found to be less than one microinch R.M.S. This modification effectively solved the seal integrity problem as related to groove geometry and surface finish.

Another area that affected seal integrity was the contamination of the bearing fluids and of the mercury itself.

The bearing fluids 1-OCTENE and 1-HEPTENE (referred to hereafter as octene and heptene) are hydrocarbon fluids with a double bond in the molecular chain. The double bond is relatively unstable so that gradual decomposition of the fluid occurs, particularly if the fluid is exposed to air or moisture. The resulting peroxides react with mercury to form mercuric oxide precipitates which can either clog the bearing gap or, in the limit, deplete the mercury and diminish the preload on the mercury to the point where differential pressure sealing capability is lost. The peroxides were found to be present even in chromatographically prepared fluids as received from the supply house. The fluids are purified and cleansed of peroxides and water by filtration through an 18 inch column of activated alumina. This procedure should take place preferably

immediately before the bearing fluid fill procedure or as part of the fill procedure. Another fluid which was used in this program was n-heptane (referred to as heptane) a straight chain hydrocarbon fluid with no double bonds. No evidence of peroxide formation of oxide precipitates were found when mercury was shaken together with a quantity of heptane. This held true even for heptane samples with no preparatory cleansing and with heptane samples in stock for a long period of time. The heptane samples were filtered through activated alumina, nevertheless, as a precautionary measure to remove trace quantities of peroxide contaminates and water, if present. The outstanding stability of the heptane and its value of viscosity intermediate between that of octene and heptene resulted in heptane being the preferred bearing fluid during the late stages of the program.

The mercury seals were formed from triple distilled mercury available from Bethlehem Apparatus Corporation. The impurity contents are certified to be less than a few parts per 100 million. After seal formation, the seals are exposed to the ambient atmosphere and may become contaminated. In order to inhibit this, the bearings were kept in sealed nitrogen filled containers for storage purposes. In addition, various cover fluids were injected into the seal gaps to protect the seals from the ambient atmosphere and to inhibit vapor pressure losses from the mercury seal. DC750 silicone fluid appeared to be a reasonably good cover fluid although no fluid appeared to be completely satisfactory in a running bearing. Here, the problem of

evaluating cover fluid effectiveness was masked and complicated by dynamic excitation and mixing of the fluids and mercury if a sizeable dynamic or unbalance existed in the test setup. Despite the difficulties in obtaining a definitive result, it is recommended that cover fluids be considered for use in future applications using mercury seals.

4.0 BEARING TESTS

4.1 BEARING LIFE

Considerable effort was expended to define this very important parameter. Two major test cycles were performed. The first test ran for slightly more than 30 days in January and February, 1975. Its purpose was merely to demonstrate life rather than to provide insight into the various measurable properties that could be considered. As a result, the associated instrumentation measured test conditions only, ambient pressure and temperature, motor speed and motor power. The second life test took place in March, 1975. At this time the instrumentation had been refined to incorporate mercury seal pressure, bearing lubricant (Heptane) pressure and bearing cartridge temperature as well as those measured in the first sequence.

A summary of the two life tests, experimental set up, results and observations is given in the following paragraphs.

- Test 1

The experimental set up consisted of a liquid bearing assembly mounted in a modified Gyroflex Gyro housing and motor package. Both mercury seals were formed and pinched off without instrumentation. The lubricant tube was not pinched off but terminated in a U tube sealed with mercury. Its pressure was not

monitored. Motor voltage and current were monitored, as was wheel speed. The entire test assembly was placed in a 12 inch bell jar and maintained at a pressure of 235 mm Hg of helium. (See Figure 3)

As the purpose was to demonstrate life, testing was confined to observations of running conditions. Wheel speed, motor current, voltage and phase angle were periodically monitored. After accumulating 840 hrs. of running time the life test was terminated.

At this point it was planned to disassemble the bearing and subject the lubricant and the mercury seal material to gas chromatographic and spectrographic analysis respectively. However, before this was done, two important tests were performed.

First, starting power was measured. Then the bearing assembly was vibration tested. The former test is self explanatory. In the latter, the bearing assembly was exposed to 6 sweeps at 1g peak from 100 Hz to 3000 Hz each of 4 minute duration. The rotating mass was 32 grams. The direction of vibration was along the spin axis. No effect was observed by instrumentation measuring the motor current.

Motor current is a moderately sensitive measure of bearing displacement. Bearing power dissipation is a non-linear function of gap.

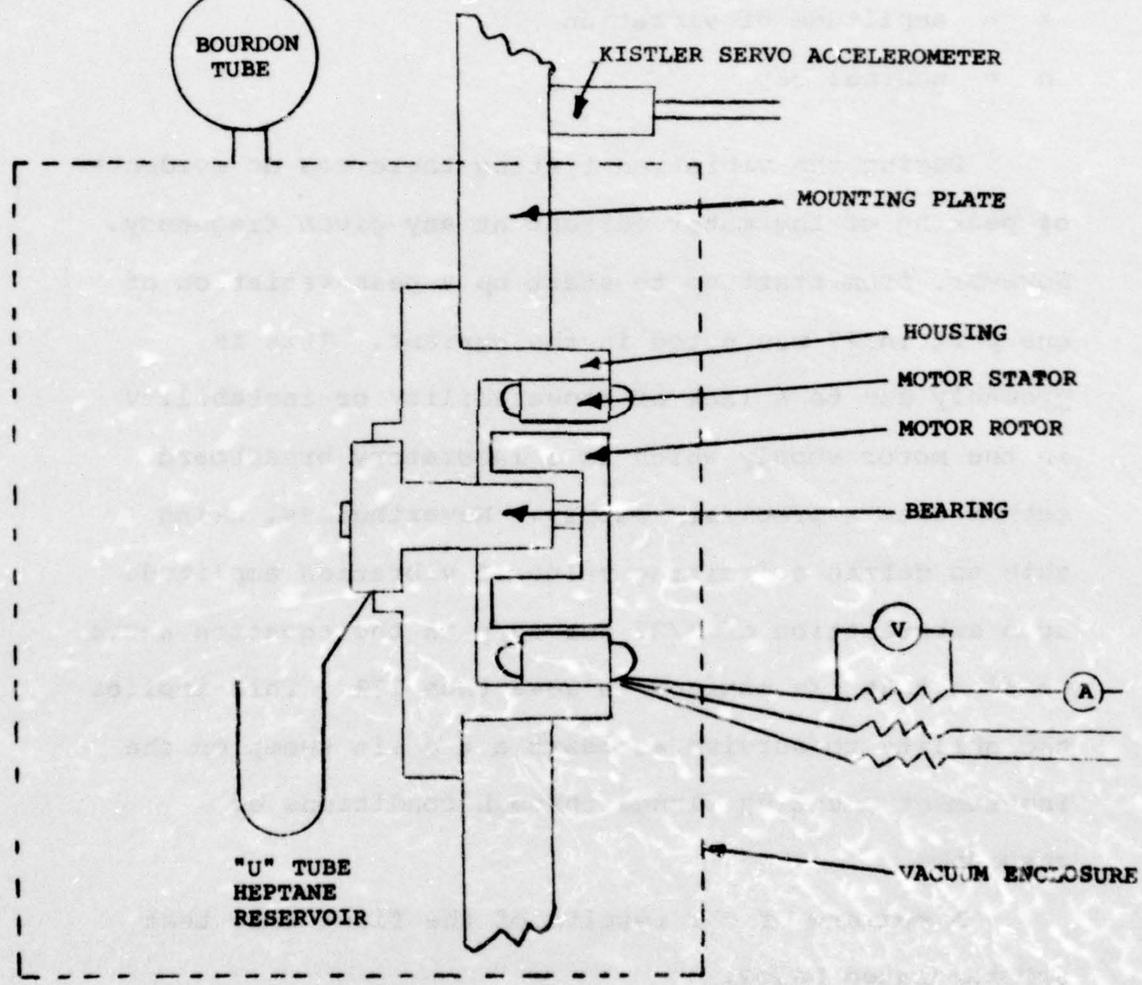

Figure 3 Life Test No. 1 Schematic

It can be shown for a self acting thrust plate pair such as in the liquid bearing

$$\frac{\Delta p}{p} = \frac{1}{2}\left(\frac{\varepsilon}{h}\right)^2$$

Δp = the power consumption increase resulting from vibration

p = nominal power consumption

ε = amplitude of vibration

h = nominal gap

During the vibration testing there was no evidence of peaking of the motor current at any given frequency. However, from start up to start up a peak variation of one part in 27 was noted in the current. This is probably due to a lack of repeatability or instability in the motor supply which is a laboratory breadboard rather than a precision supply. Nevertheless, using this to define a limiting value of vibration amplitude upon substitution of 1/27 for $\Delta p/p$ in the equation above we find that ε/h maximum is less than 27%. This implies the ability to survive at least a 4 g sin sweep on the instrument mounting flange through conditions of resonance.

A summary of the results of the first life test are tabulated below.

Results of Life Test 1

Parameter	Result
Running time	840 hours
Lubricant	Heptane
Atmosphere	235 mm Hg of Helium
Mercury Preload	8 psi
Heptane pressure	1 psi
Temperature	70°F
Self vibration	30 milli g
@70°F running power	2 watt
@70°F start power	4.2 watt
Residual unbalance	Constant
Analysis of disassembled bearing	
Heptane (chromotographic)	No contamination detected
Mercury spark (spectographic)	No contamination
Vibration Resistance	less than 27% eccentricity for 1g sin input 100 to 3000 Hz in 4 minutes

Although the first life test revealed considerable information on the performance and longevity of the liquid bearing, it had the major defect of being a go - no go type test. It offered no insight into conditions within the bearing. In particular it was not possible to sense small changes that ultimately would lead to failure. As a result, a second test was run. It was more sophisticated than the first in that direct measurements on a continuous

basis were made of the pressure in the heptane, pressure in one mercury seal and temperature of the bearing cartridge.

· Life Test #2

The test set up included a liquid bearing cartridge assembled in a Gyroflex Gyro motor and housing assembly. One mercury seal was pinched off in the normal manner. The other seal was instrumented through a Pace P-1 pressure transducer. The heptane line was also instrumented with a Pace, P-1 pressure transducer. Both pressure transducer outputs were monitored continuously on a chart recorder. See Figure 4.

The heptane system was sealed in a reservoir which contained a bellows to accommodate thermal expansion effects.

A copper constantan thermocouple was bonded to the bearing cartridge with adhesive. Its output was continuously processed through a differential voltmeter and a chart recorder.

The bearing assembly was maintained in a bell jar in a reduced pressure of 4.6 psia of Helium.

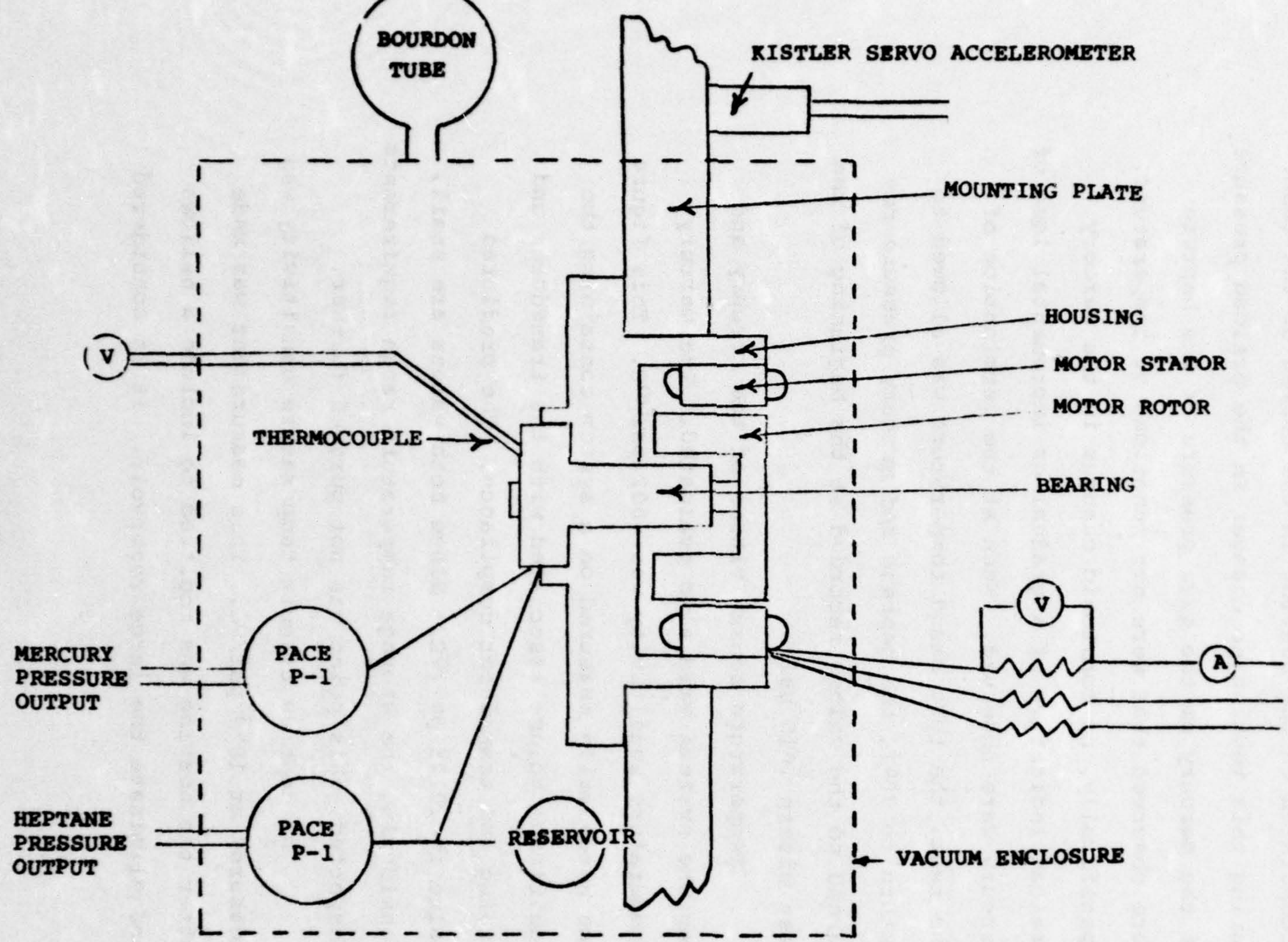

Figure 4 Life Test No. 2 Schematic

RESULTS OF SECOND LIFE TEST

Approximately 400 hours of running time were accumulated on the bearing in the course of the test. During this testing no changes in the preload pressure of the mercury or the gage pressure of the heptene were observed that were not functions of temperature. Specifically, no monotonic changes in the mercury preload indicative of a gradual or incremental loss of mercury were observed. When at the termination of the test, the test stand temperature was allowed to return to 70°F, the heptane and mercury pressure returned to the values recorded at the beginning of the test within .015 psi.

Temperature sensitivities of the mercury and heptene systems were also evaluated. The mercury temperature sensitivity was .002 psi/°C. This figure was necessarily measured on a system containing the additional volume associated with the transducer and tubing and transducer compliance. The predicted value is .0125 psi/°C. Since both values are small, considering the storage temperature range requirements respected, this point was not pursued further.

The heptane circuit temperature sensitivity was measured at 10^{-3} psi/°C. This measurement was made after the bearing was modified to include a bellows and eliminate the large reservoir. It is considered

feasible to incorporate a bellows in the gyro for thermal expansion purposes so this is not an extraordinary modification. Here again the volume of liquid in the system was larger than would be expected in a gyro, but since the product of the coefficient and the anticipated temperature range is low, no further effort was considered worthwhile.

Other measurements made during this test include:

- Bearing operating temperature rise. A typical temperature rise upon starting the motor was roughly 22 deg. F. This was observed many times and thus is a representative number.

- Running power stability. One main reason for utilizing a liquid bearing is that its power dissipation is constant at a constant operating temperature. Ball bearings on the other hand are subject to instantaneous power dissipation changes. These effects, referred to as "power jags" are evaluated by a sustained monitor of motor current. Such a monitor was maintained and no discontinuous current variations were measured to a resolution of .1 ma.

· Self Vibration

A Kistler model 305A Servo accelerometer was mounted adjacent to the bearing assembly. The vibration level was monitored for stability. No changes of any consequence were observed.

· Minimum start voltages and bearing power consumption

The minimum voltage to produce breakaway and levitation was measured for spin axis horizontal and spin axis vertical. In each case it was found that 4 volts would usually produce levitation, but 6 volts were required for reliable starting. Bearing power consumption was unchanged from the first test.

· Fluid Pressure Effects

It had been postulated that measurable pressure changes should occur in both the heptane and mercury systems upon starting or stopping the bearing. None were observed in the heptane system. In the mercury system small variations in the order of .001 psi p-p have been observed at wheel speed during run down to a stop. Pressure variations of the same order of

magnitude can be produced by pushing the shaft back and forth along its longitudinal axis. It is felt that the two effects are related. A likely cause of axial motion would be a lack of squareness in the thrust plate and housing.

High frequency effects were sought in both the mercury and heptane circuits. None were found.

SUMMARY OF RESULTS - LIFE TEST 2

Parameter	Result
Running time	400 hrs
Lubricant	Heptane
Atmosphere	330 MM helium absolute
Mercury preload	10 psi
Heptane pressure	Less than lpsig
Temperature	$70^{o}F$ nominal
Self Vibration	30 milli g
Running power	2 watt Typical
Start power	4.2 watt
Mercury Temperature Coefficient	.002 psi/^{o}C
Heptane Temperature Coefficient	.001 psi/^{o}C
Nominal Bearing Temperature Rise upon start up	$22^{o}F$

4.2 LOAD DEFLECTION TESTS

Both the radial and axial load carrying capacity of an early model of the liquid bearing were evaluated. Since the liquid bearing must rotate to have stiffness fixturing was designed that applied loads and measured displacement without mechanical contact with the shaft. Two setups are required to accomodate both radial and axial loads and displacements. They are described schematically in Figures 5 and 6.

Referring to Figure 5, the basic test fixture is a gyro housing in which a liquid bearing assembly has been installed. The liquid bearing is fitted with 0.25 inch diameter shaft extensions that run true to the bearing journal to within 0.0002 inch TIR. The gyro housing is mounted rigidly to a plate. Also attached rigidly to the plate are a capacitance displacement transducer (Photocon Research Products PS 600) and a pressure tube. The former is a capacitance bridge type of displacement transducer with a built in micrometer calibration feature which makes it very useful for static measurements. The latter is simply a 0.202 inch diameter tube located colinearly with and roughly 0.005 inches separated from the shaft extension.

A calibration for the pressure tubes is given in Figure 7. Calculated force vs. pressure curves are given for both one tube alone and two tubes acting in concert as in the radial load deflection fixture. A measured curve for 2 units is also given. The agreement

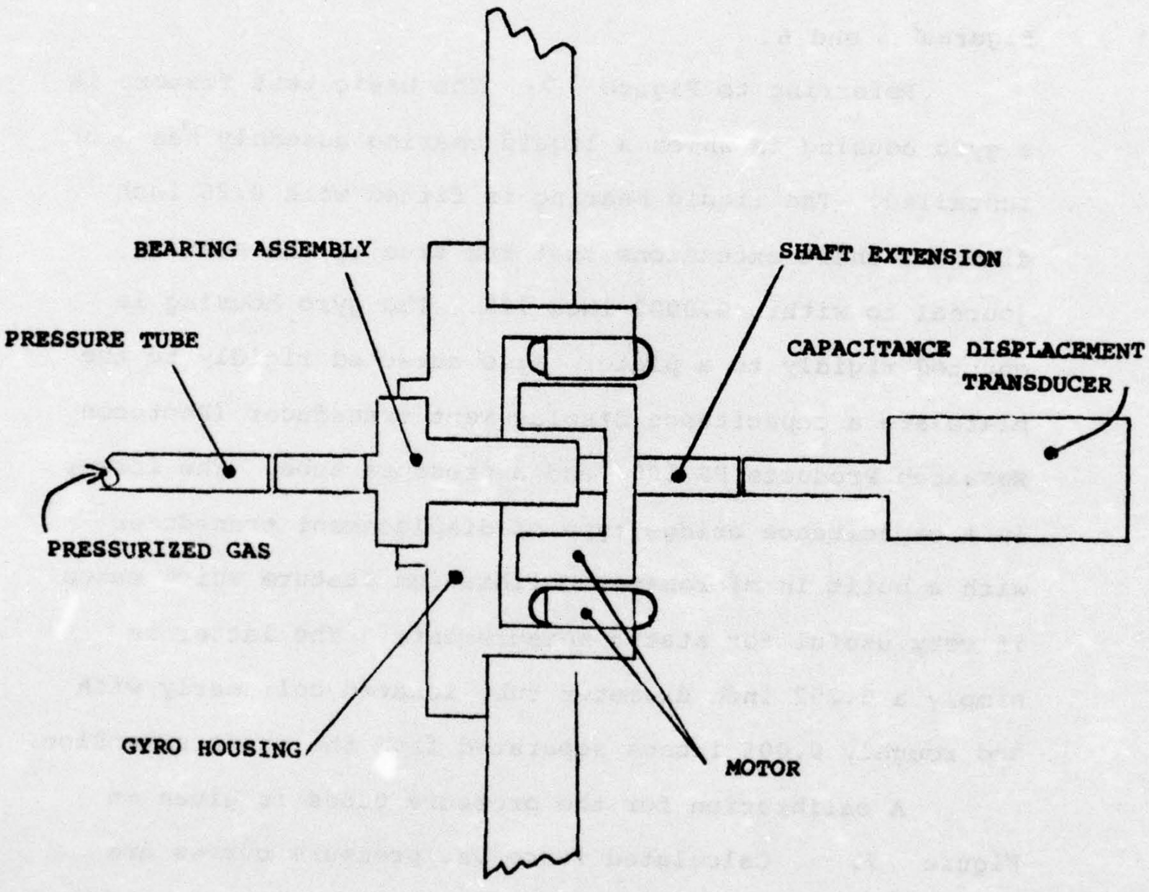

Figure 5 Schematic of Axial Load Deflection Fixture

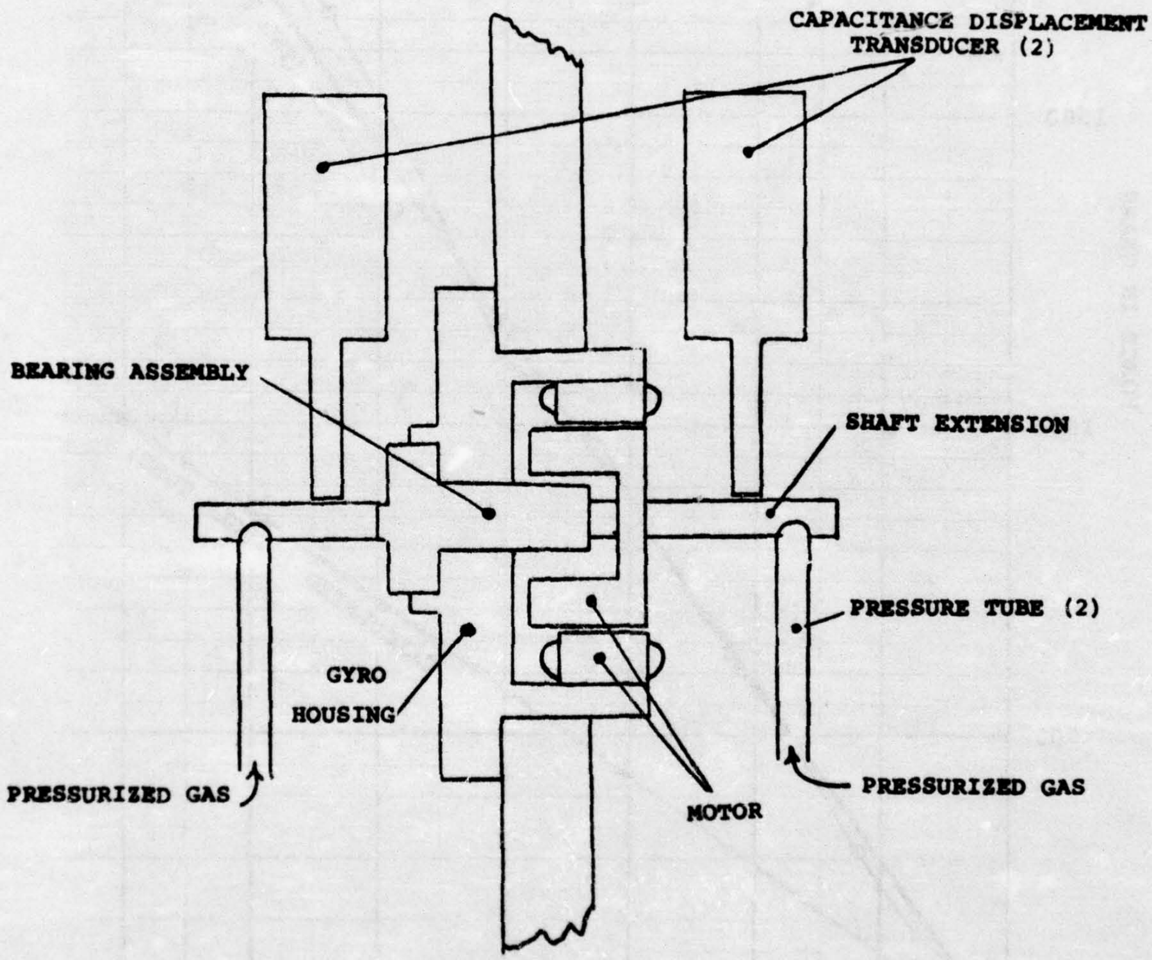

Figure 6 Schematic of Radial Load Deflection Fixture

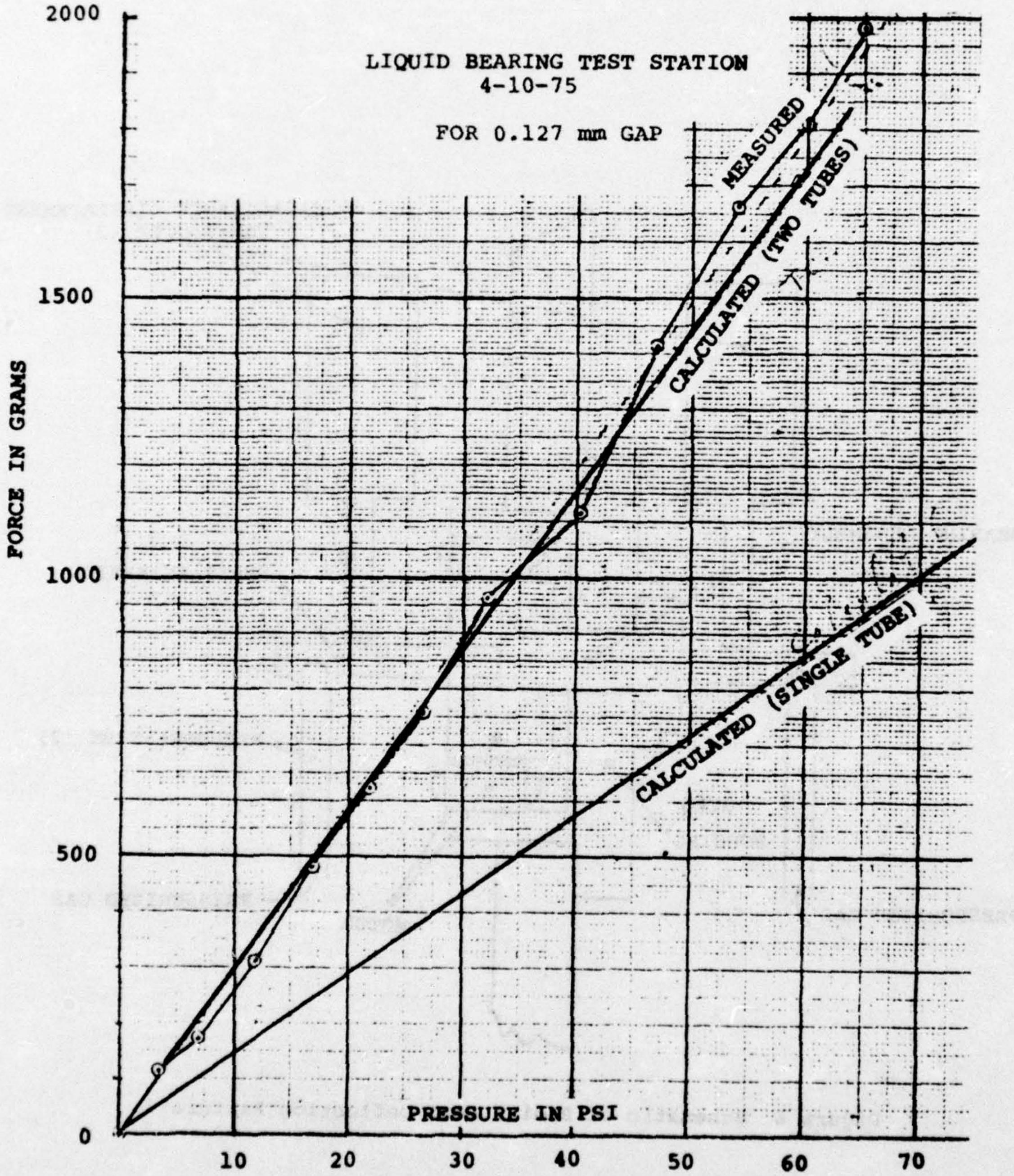

Figure 7 Load Calibration Curve

is quite good, particularly at lower pressures. This comes about because the only error mechanisms in this type of force producing device are pressure drops between the pressure sensor and the end of the tube, and flow effects around the annular exit formed by the tube and the shaft extension. Both effects are small at low pressures. Another advantage of this approach is that as long as the pressures are not very high, the force is independent of the clearance between the tube end and the shaft extension.

Referring to Figure 6 it can be seen that the radial load deflection fixture requires two pressure tubes and two sensors. This becomes necessary because the center of suspension of the bearing is not accessible to the pressure tubes. A force applied anywhere else will produce an angular displacement as well as a linear displacement.

The configuration of the pressure tubes is slightly altered in the radial load deflection fixture. In order to maintain a close clearance between the tube and shaft extension the tubes are not terminated square but curved to an 0.005 inches greater than the radius of the shaft extension. This is illustrated in Figure 8.

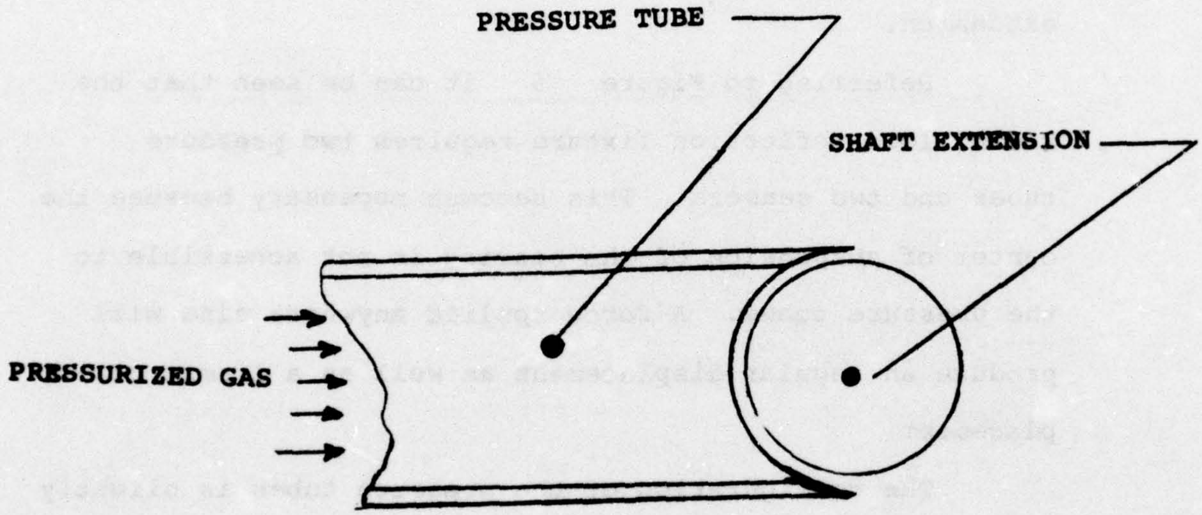

Figure 8 Radial Load Deflection Fixture Detail

4.2.1 Axial Stiffness

Axial spring rate measurements were performed on two bearings with different groove geometries. The principal features of these thrust plates are compared below where "old" design refers to the original Kearfott design and "improved" refers to the thrust plate fabricated in the course of this program.

	OLD DESIGN	IMPROVED DESIGN
O.D.	.480 inch	.480 inch
I.D.	.244 inch	.244 inch
Groove I.D.	.404 inch	.324 inch
Gap	150 micro-inch	150 micro-inch
Temperature	$90^{o}F$	$90^{o}F$
Lubricant	n-heptane	n-heptane
Viscosity	.52 cs.	.52 cs.

The principal difference between the two thrust plate designs is the increased length of the grooves in the new design. Since the groove I.D. of the old design was too large, a low value of stiffness was expected. The predicted stiffening effect of the new design due to a decrease in groove I.D. was verified in subsequent testing.

Axial spring rate testing was performed in the fixture shown in Figure 5, which applies axial forces against a bearing adapter with a single pressure tube. A comparative plot of the spring rate tests for both designs is shown in Figure 9.

A high degree of confidence in the precision of the test results is based on the fact that this particular set up minimizes the effects of error sources such as fixture compliances, angular displacements, transducer drift, and thermal effects. The simplicity of the set up made this test the easiest to perform of all three stiffness measurements made. Since a test series over the total force range could be performed in a matter of two to three minutes, drift effects in the single transducer were practically negligible. Also, the cold expanding gas from the pressure tube could not be deflected against the transducer. This factor reduced thermal effects as a source of error.

The spring rate of 14,700 lb/inch for the old design bearing is low. It is seen that the improved design with longer thrust grooves gives an axial sprin_ rate of 27,500 lb/inch, or an increase in stiffness of almost two to one.

Re-examination of various analytic solutions found in the literature leads to a predicted axial spring rate of approximately 50,000 lb/inch for thrust plate bearings

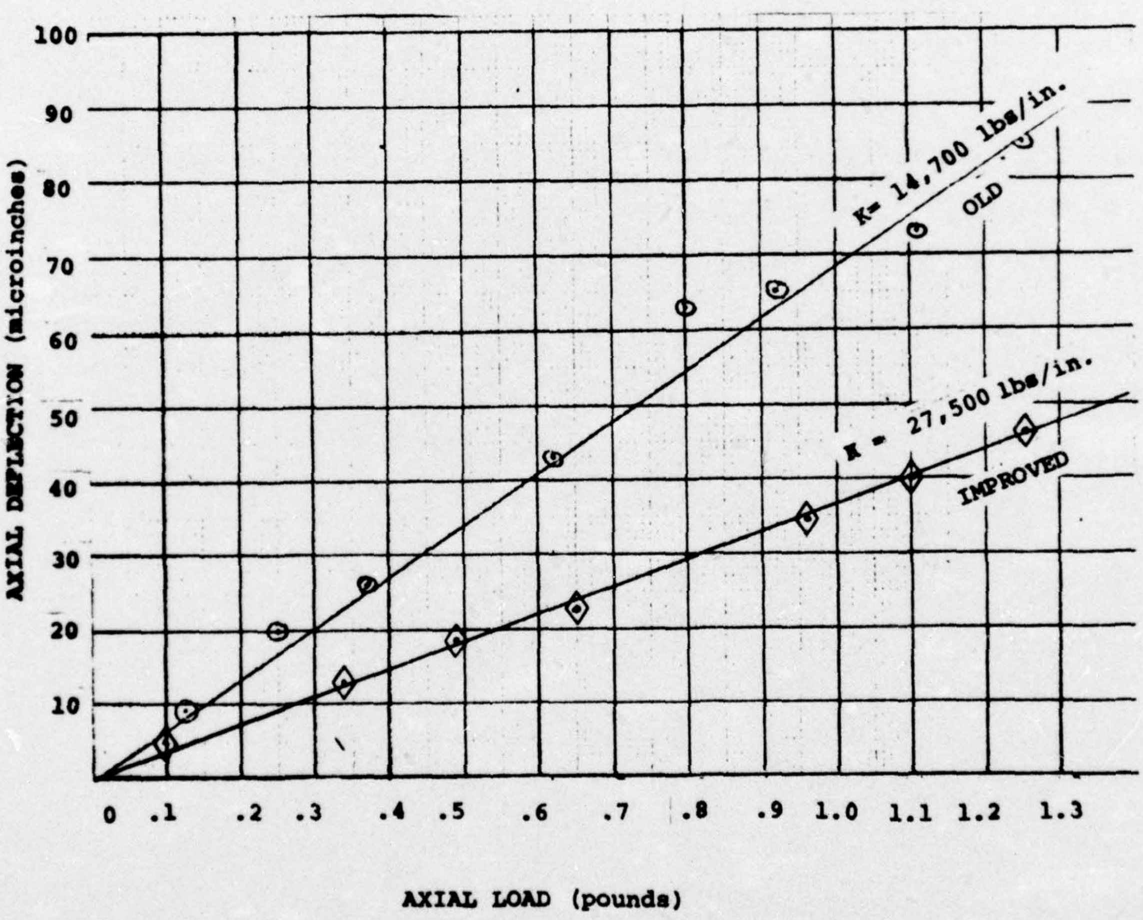

Figure 9 Axial Deflection vs Load

of the same envelope and power consumption but with optimized groove configurations.

It is expected that a bearing with optimized groove configurations will be available for testing in the near future. It is planned to verify the validity of the analytical solutions with axial spring rate testing of the optimum design.

4.2.2 Journal Bearing Stiffness

The journal bearing tested had the following geometry:

Length	.665 inch
Diameter	.250 inch
Gap	150 micro inch
Groove Configuration-3 equal sections	

Both outer segments contains inward pumping grooves.
The central section is smooth.

Temperature	90°F
Lubricant	Heptane
Viscosity	0.52 CS

The testing was conducted in a fixture similar to that shown schematically in Figure 6 . Several special requirements exist in this particular test that forced the test techniques to differ from that used in the axial load testing.

First, as a result of the bearing geometry, it is impossible to apply force loads on the center of bearing suspension. Therefore, as a corollary, whenever the bearing is loaded radially, significant angular loads are inadvertently applied. Second, it is extremely difficult to maintain stable thermal conditions in the presence of large quantities of expanding gas issuing from the pressure tubes. In order to achieve meaningful data in the pressure of these conditions, the following test procedure was devised:

The testing was performed about discrete loads rather than over a series of increasing loads as in the axial load testing. For example, the test for which the raw data in Figure 10 was taken was run with a pressure of 15 psi on one pressure tube and pressures ranging from 9 to 21 psi on the other pressure tube.

In order to be able to extract useful information from the raw data, it is essential that the thermal effects be stabilized to the point that the zero pressure readings before and after the test agree within 10 micro inches. As can be seen this condition is fulfilled in the attached data. This was accomplished by operating the pressure tubes at the nominal pressure (15 psi) for approximately 5 minutes. Then when the temperature was reasonably stabilized the pressure was turned off while the chart paper zero was reestablished. Then the test is run.

The sample data is calibrated at 5 micro inches of journal displacement per division. One pressure transducer is fixed at 15 psi .The pressure in the other transducer is written on the chart next to its associated displacement. Each increment of pressure, one psi is equivalent to a force increment of 14,000 dynes (.031 lb) and a torque increment of over 35,000 dyne cm. The data reduction is accomplished by computing the angular and linear displacement of the centroid of the bearing. A spring rate was computed from the load, and linear deflection at that combination of pressures that gave the smallest angular displacement. This process was repeated at four

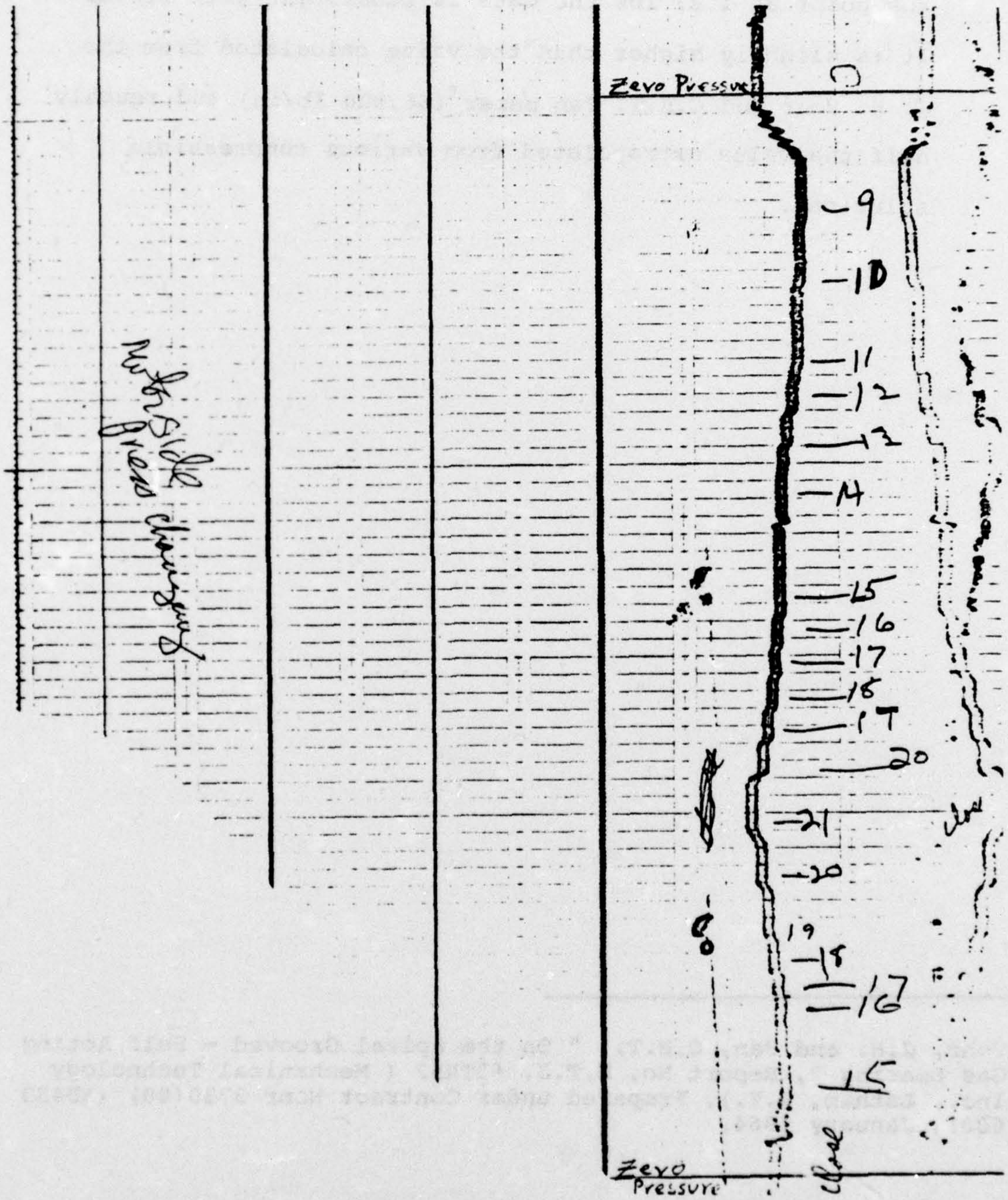

Figure 10 Journal Bearing Load Deflection Data

different nominal bearing loads covering a range of loads from 1 to 2.5 lbs (6 to 16g). The results of the tests are summarized in Figure 11. With the exception of the point at 1.87 lbs the data is consistent with itself. It is slightly higher than the value calculated from the J. H. Vohr and C.H.T. Pan paper* (66,900 lb/in) and roughly half the value extrapolated from various compressible solutions.

* Vohr, J.H. and Pan, C.H.T., " On the Spiral Grooved - Self Acting Gas Bearing ", Report No. M.T.I. 63TR52 (Mechanical Technology Inc., Latham, N.Y.), Prepared under Contract Nonr 3730(00) (AD433 600), January 1964.

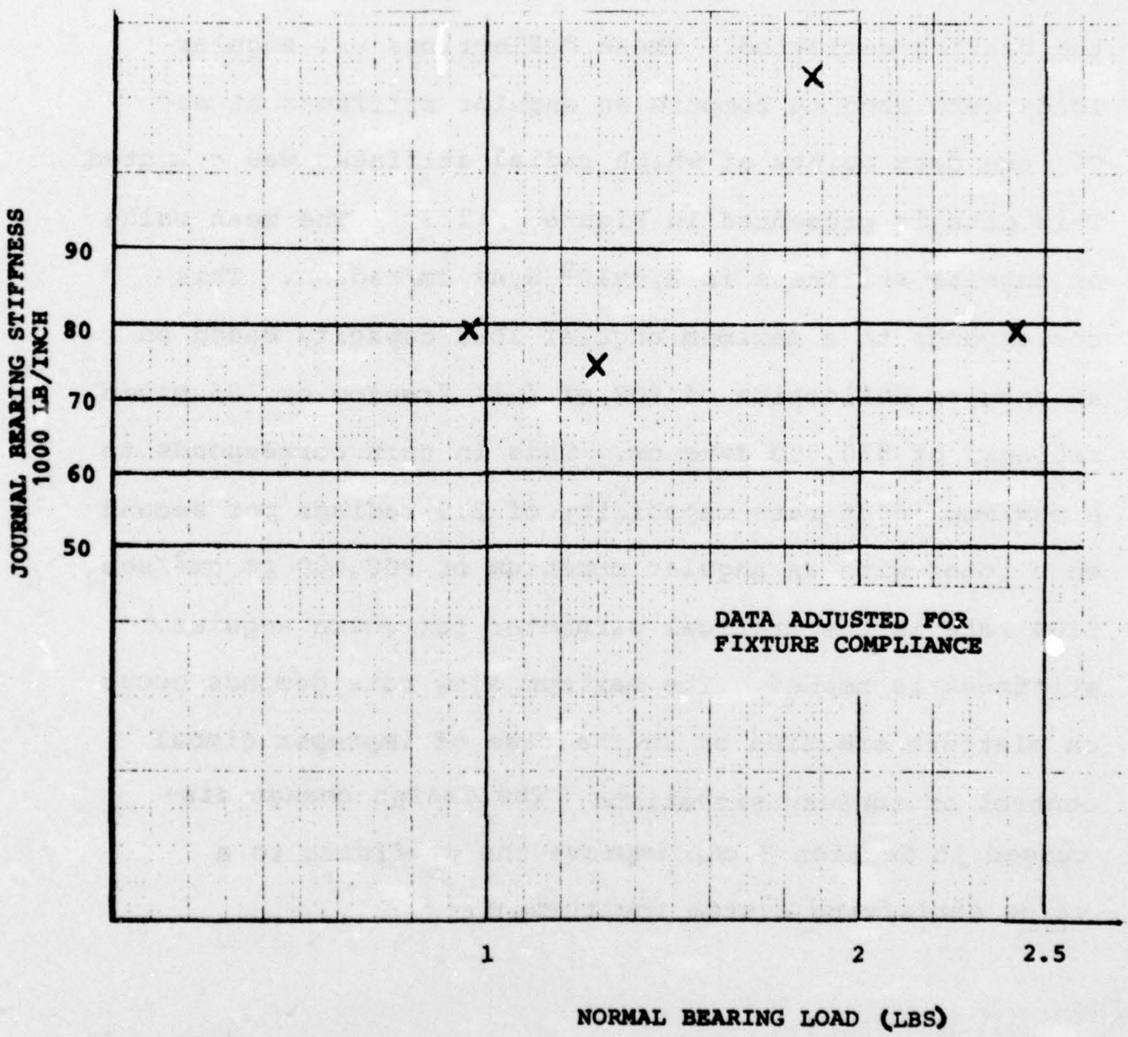

Figure 11 Journal Bearing Stiffness vs Load

4.2.3 Bearing Angular Stiffness

The bearing angular stiffness is a natural outcome of the method of measuring the journal bearing stiffness. While the pressure in one pressure tube was maintained constant , the pressure in the other tube was varied over a range of $\pm$ 5 to $\pm$ 7 psi about the nominal pressure. As the pressure varied, large angular inputs occurred, and the bearing deflected. These deflections vs. angular loads were used to compute an angular stiffness at each of the data points at which radial stiffness was computed. This data is presented in Figure 12 . The mean value of angular stiffness is 2.6×10^9 dyne cm/radian. This corresponds to a maximum angular load capacity based on an angular deflection of 80% of full freedom or 195 micro radians, of 510,000 dyne cm. This in turn corresponds to a maximum input rate capability of 2.5 radians per second on a rotor with an angular momentum of 200,000 gm cm^2/sec. Slew rate is the critical parameter for which angular stiffness is needed. The maximum slew rate demands occur on platform erection or in the case of improper gimbal control or runaway situations. The design change discussed in Section 3 can improve the stiffness to a value satisfying system requirements.

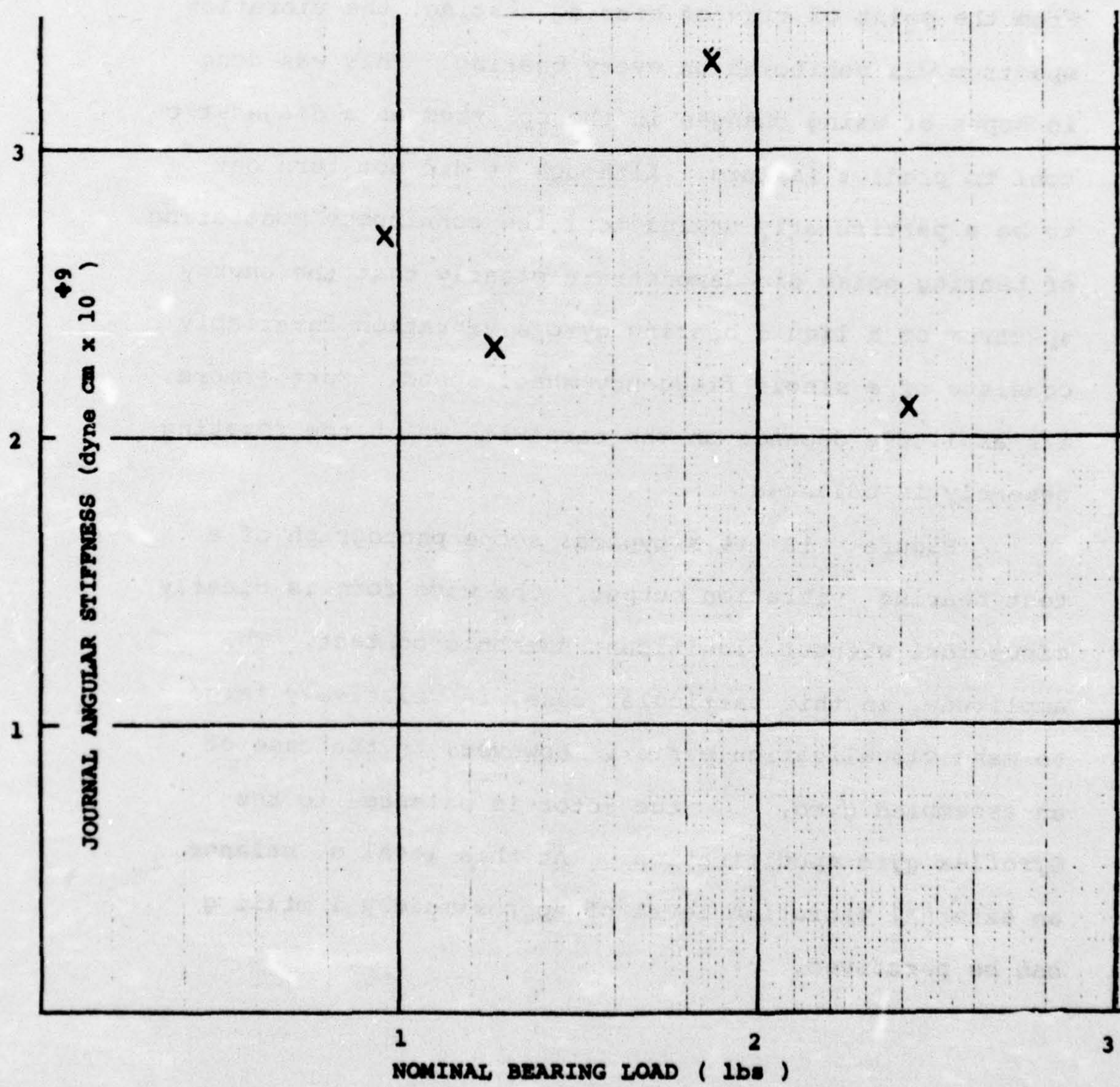

Figure 12 Journal Bearing Angular Stiffness vs Load

4.3 BEARING NOISE EVALUATION

One of the main reasons for developing a liquid bearing for the spin bearing in the Gyroflex gyro is its inherent low noise properties. It was felt (later proven correct) that the mechanically simple structure of a dynamic spin bearing would have a simple noise spectrum. From the point of view of bearing testing, the vibration spectrum was monitored on every bearing. This was done in hopes of using changes in the spectrum as a diagnostic tool to predict failure. Although it did not turn out to be a particularly useful tool,the continuous monitoring of bearing noise did demonstrate clearly that the energy spectrum of a liquid bearing gyro's vibration invariably consists of a single frequency,wheel speed. Furthermore, its amplitude depends on the care with which the rotating assembly is balanced.

Figure 13 is a typical scope photograph of a test bearing vibration output. The wave form is clearly sinusoidal without significant harmonic content. The amplitude, in this particular case, is relatively large to make visualization easier. However, in the case of an assembled gyro, the rotor is balanced to the Gyroflex gyro specification. At this level of balance, an external vibration level of approximately 1 milli g can be perceived.

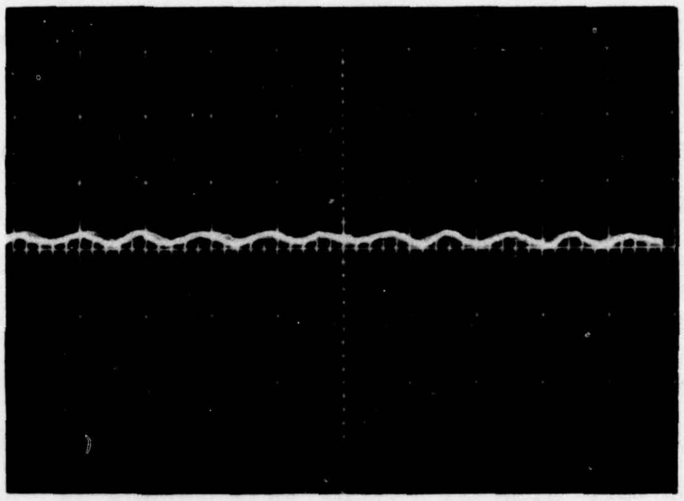

Figure 13 Liquid Bearing Self Vibration

4.4 BEARING POWER STABILITY

The constancy of bearing power dissipation is a principal source of gyro randomness. As the power dissipated in the bearing varies, power into the motor changes upsetting the thermal equilibrium of the gyro.

In the ball bearing Gyroflex gyro, power variations are manifested in the form of current "jags". Current jags are abrupt changes in current resulting from slight changes in the lubricant distribution in the races or from a ball following a slightly different path in the race. The resultant "jags" are in the order of one milliamp and correlate with bias shifts in the order of 0.003 deg/hr.

Theoretically the liquid bearing has no mechanism for producing current jags. In practice, tests in the past have demonstrated excellent current constancy. In a 16 hr. current stability run made for this program maximum current variation of $\pm$ .25 milliamps was observed. These variations were gradual rather than abrupt and undoubtedly tracked the laboratory temperature variations over night. The gyro body was not temperature controlled and thus was subject to the influence of the ambient temperature. No abrupt changes or jags were observed with a resolution of 0.1 ma. This is at least 10 times better than the ball bearing gyro.

Another important source of power variation is bearing loss variations with input acceleration. The

journal bearing losses as a function of load were measured during load deflection testing. Using a power meter with a resolution of 0.05 watts, no change in power level was observed for inputs up to 19g, at which point a monotonic increase occurred. This level of power stability greatly exceeds the requirements for aircraft navigation system gyros.

In addition to earlier tests of current jags on a liquid bearing assembly, a current jag test was performed on a bearing which had been subjected to a 4 g sinusoidal axial vibration test. The results to this test were consistent with earlier results. The duration of the test was 2½ hours at ambient room temperature.

The data showed that the initial thermal gradient resulting from a cold start without external temperature control gave a 1 ma. monotonic change in current after approximately 20 minutes of operation. For the remaining 130 minutes of testing, a very smooth current trace was obtained with occasional variations of the order of ± 20 to ± 30 microamperes. A monotonic drift in the absolute value of motor current amounting to 0.5 ma. was noted during the 130 minute observation period. This can be attributed to the variation in ambient room temperature.

The results of this test clearly demonstrated and reaffirmed the noise free character of the liquid bearing design.

4.5 BEARING VIBRATION RESISTANCE

The gyro vibration requirements are set forth in Figure 14 . Bearing tests designed to assess the ability of meeting these objectives were conducted in a fixture described schematically in Figure 15 . The vibration fixture is a cylinder fitted with mounting surfaces to adapt a gyro housing to a shaker head. The cylinder is also fitted with mounting holes to accommodate probes for a Mechanical Technology Incorporated Photonic Sensor. This is a fiber optics device that measures small displacements by the variation in the ability of a fiber optics bundle to transmit light to itself off a displaced reflecting surface. The gyro assembly is fitted with a dummy rotor machined true to the shaft.

The experimental procedure used was to search the bearing response from 100 to 3000 Hz at a fixed acceleration level. Initial tests were run with the bearing in the vertical plane at a 1g input. No discernable change could be observed at this level so the input level was increased to 4 g's. Again no response was identified as a fixture resonance was observed. The experimental sensitivity was such than an amplification of 1.1 times was detectable.

An analysis based on the paper by Richardson, et. al.* indicated the reason for the low amplification. They predict a damping coefficient of 62.9 lb. sec/in. This means that for the proposed thrust plate where a spring rate of 40,000 lb/inch is expected the damping force exceeds

*Richardson, H.H., Griffen, W.S., Yamami, S., " A Study of Squeeze Film Damping", M.I.T. Industrial Liaison Program, Reprint No. 483, 1965.

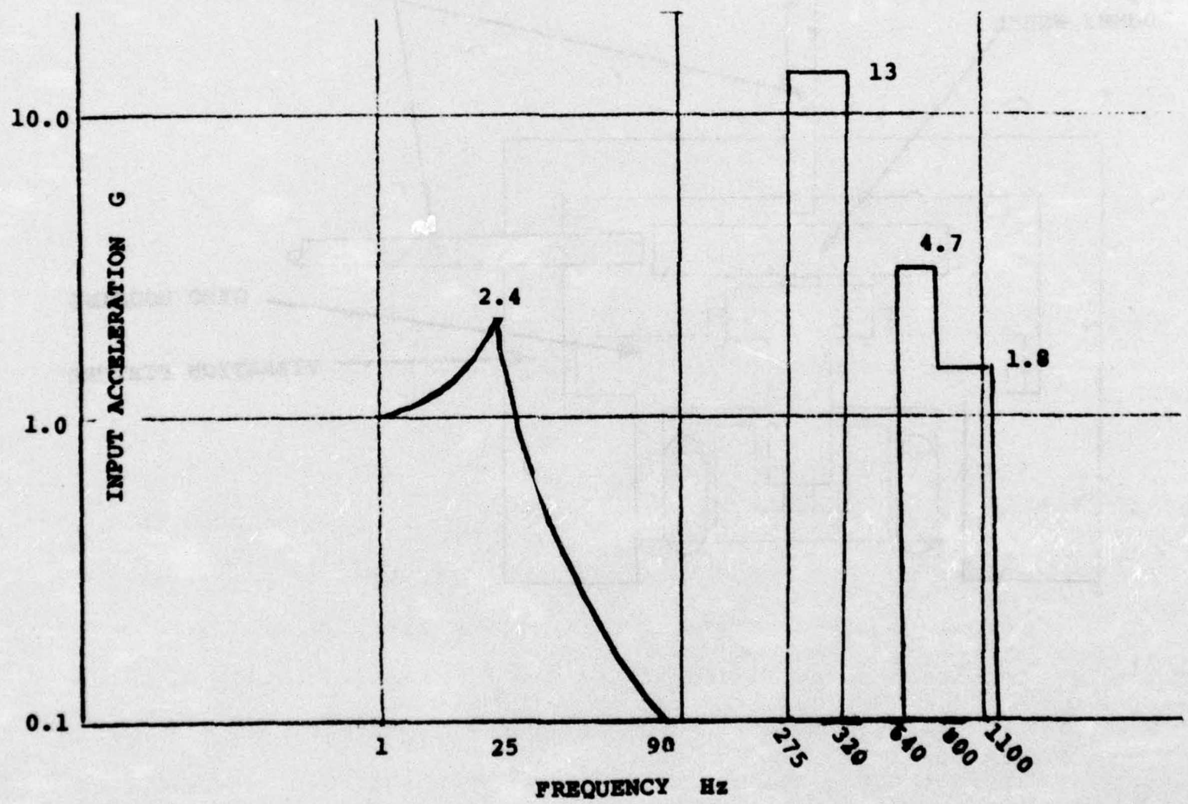

Figure 14 Typical Gyro Input Acceleration

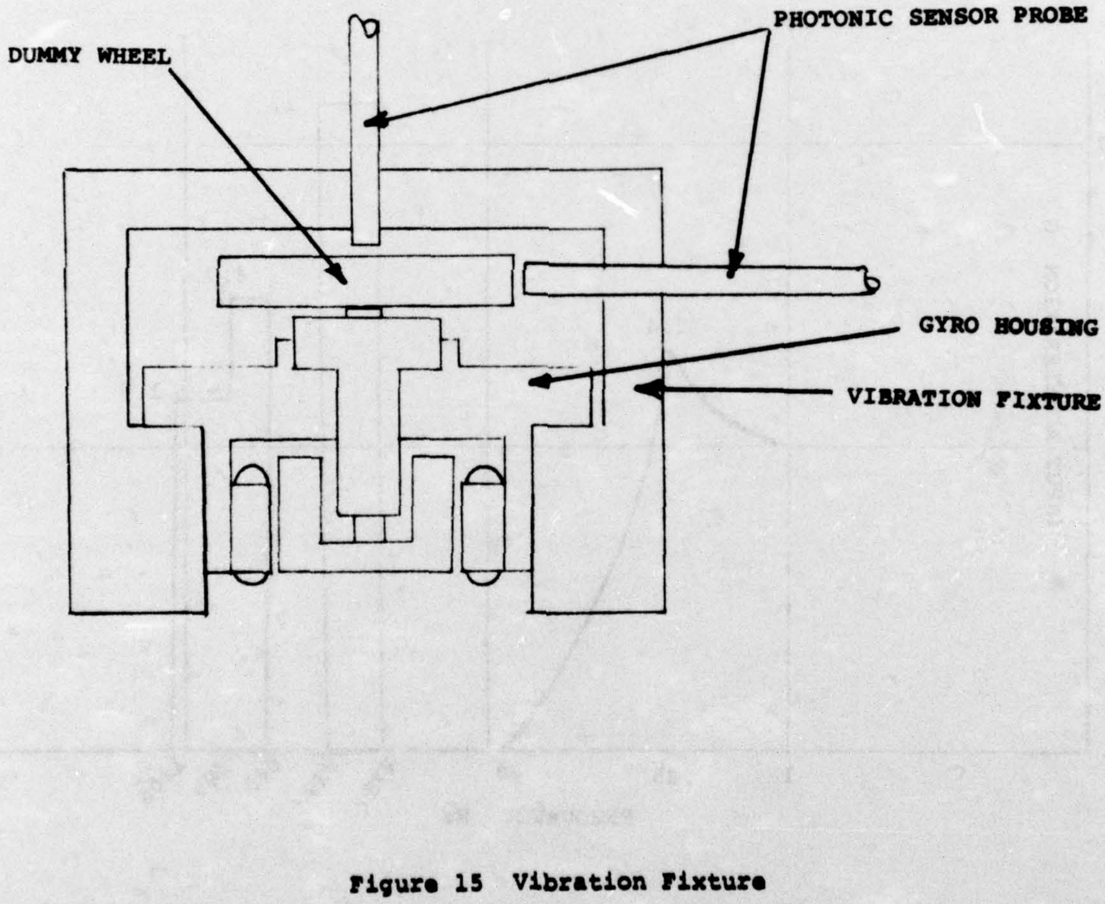

Figure 15 Vibration Fixture

the spring force at all frequencies above 100 Hz. This corresponds to a damping ratio of 4.9. Thus all motion is heavily over damped and good resistance to shock and vibration can be expected.

The experimental procedure was repeated for the journal bearing with two fixed accelerations of 2 and 4 g's over a frequency range of 100 to 3000 Hz. Calculation of the natural frequency of interest in this mode showed the number to be 3930 Hz which is above the maximum frequency tested by a factor of 1.31/1.

Deflections were corrected to show motion at the journal bearing location and are worst case because they are the combination of both linear and angular displacements.

As in the thrust plate measurements no response not identified as a fixture response was observed. The experimental sensitivity was such that an amplification of 1.4 times was detectable in the 2 g test and 1.2 in the 4 g test. Figures 16 and 17 show a plot of the response for these two tests.

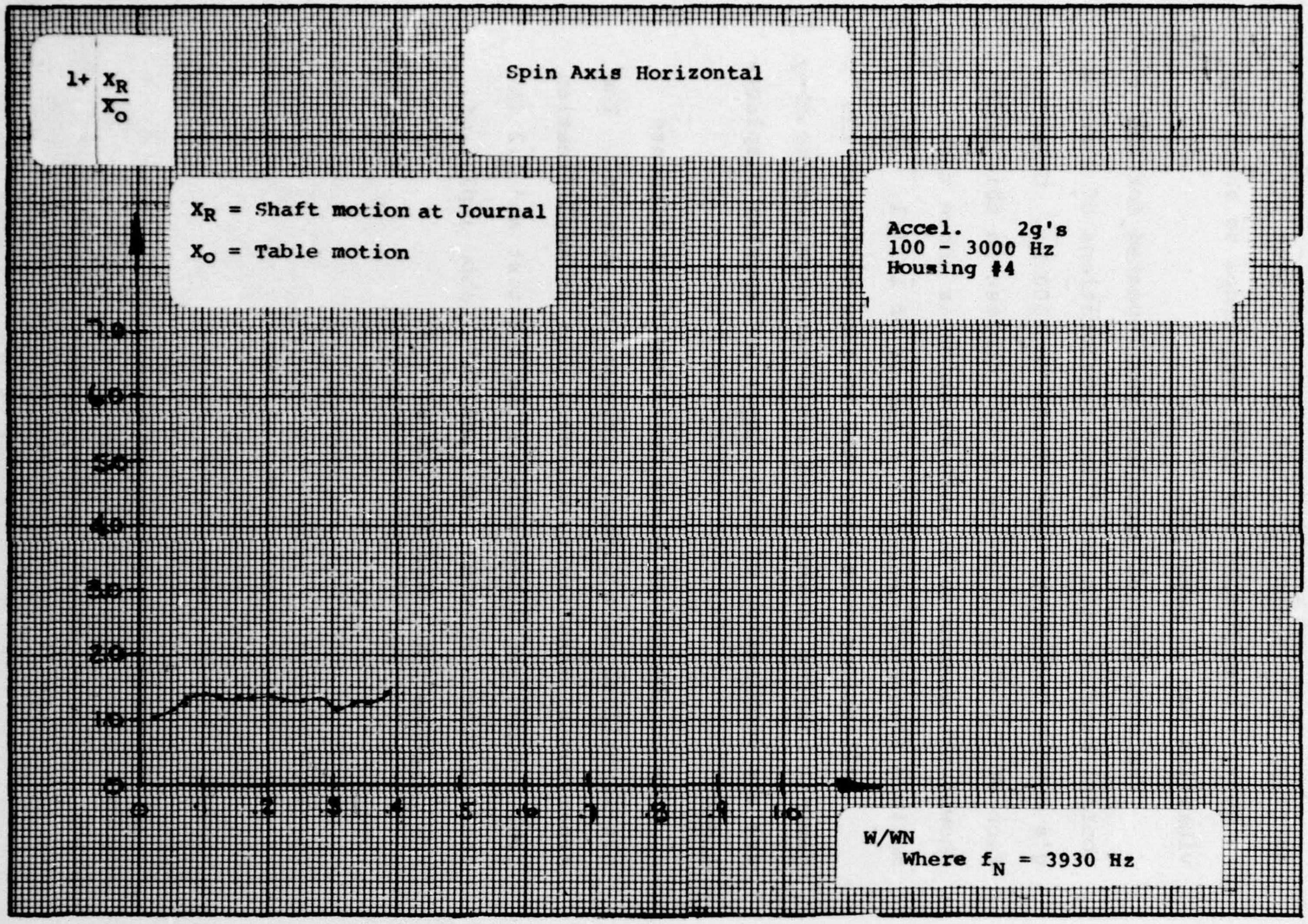

Figure 16 Vibration Response 2 g's

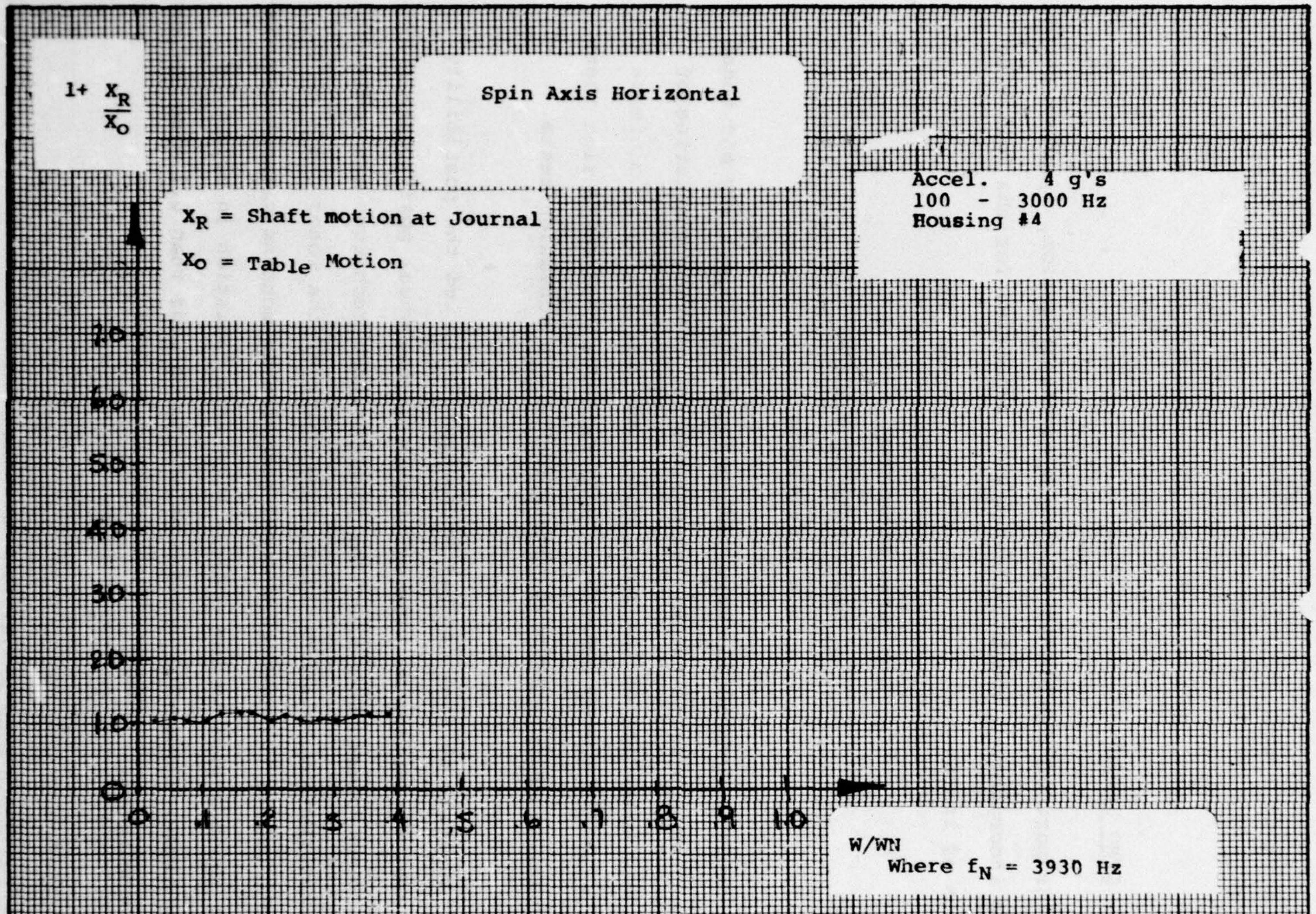

Figure 17 Vibration Response 4 g's

5.0 GYRO TESTS

As part of the Liquid Bearing Gyro Investigation, Gyroflex gyros with incorporated liquid bearings were scheduled for the following types of tests:

- Bias Stability
- Environmental Testing
- 2 N Sensitivity

Several gyro builds were made using a liquid bearing and these were then subjected to initial calibration testing consisting of dynamic balancing of the rotor and cover positioning to minimize the internal gas spring constant. Seal failures occurred during these operations which precluded completion of the scheduled tests.

Analysis of seal failure mechanisms indicated the possibility of seal disruption of high "g" inputs in the wheel positioning phase of rough dynamic balance. The rough balance technique was modified to incorporate variable spring tensioning of the rotor shaft which would permit the necessary wheel positioning motions without requireing shock inputs to the seals. With this variation in balancing technique, successful dynamic balancing to less than 0.2 dyne cm. was accomplished.

In the next step in the calibration procedure, the gyro was operated at 14,400 r.p.m. in a hydrogen atmosphere at 50°C in order to obtain data on the spring restraint attributable to the interaction of the atmosphere with the rotating wheel. The cover was then rotated so that the slightly asymmetrical gap between the cover and the wheel was properly positioned to minimize the gas spring restraint effect. In general, six to eight hours of running time was accumulated in the dynamic balance and cover position tests. Seal failures developed during the cover position test after this time. Failure analysis indicated that the seal failures were caused by the "oil-canning" expansion at low test chamber pressures of a slack diaphragm bellows attached to the heptane fill line. This allowed the internal fluid to vaporize with subsequent catastrophic bottoming out of the bearing on the next stop in the test cycle. A double diaphragm bellows with an internal gas reference pressure chamber was fabricated and successfully tested down to less than 100 microns chamber pressure. Since the test chamber pressure is usually held at a higher pressure, this was felt to be a satisfactory test procedure.

Subsequent testing of another liquid bearing gyroflex gyro was terminated with a seal failure in the cover position test.

Critical re-examination of the bearing and the test procedure indicated that the relatively low angular stiffness of the bearing design permitted seal disruption to occur in gradual steps especially during dynamic balance and during start-stop cycles. Preliminary bearing designs were conceived during the program which would give greatly enhanced angular stiffness but sufficient time was not available at that point in time to implement these designs into working hardware.

6.0 PHYSICAL COMPATIBILITY WITH THE KT-70 SYSTEM

Three areas that must be considered in the evaluation of the compatibility of the liquid bearing Gyroflex gyro with a KT system are:

- Electro/mechanical interface compatibility
- Environmental compatibility
- Performance

As will be seen below interface restrictions determine bearing environmental capability so they will be discussed together.

The electromechanical compatibility has been studied in detail in a design effort and in a study of the impact of the substitution of a liquid bearing cartridge on the magnetic properties of the gyro housing. The results of this activity define a bearing package outline that can be installed into the Gyroflex gyro body with a minimum of machining (one bore is opened slightly, and a boss is machined back to provide a reference surface). The results of the magnetic study was negative. The insertion of the hard steel bearing cartridge in the soft magnetic iron body has no effect on the pickoff scaling.

The design study defined the bearing geometry that could be fit within the Gyroflex gyro outline. It is possible to use existing design papers to predict

radial and axial load capacities for that design. However, there are no design analyses available for the angular stiffness of a spiral groove journal bearing. The bearing performance measurements, however, provided a baseline angular stiffness from which the performance of the geometry desired could be scaled. It was assumed that angular stiffness scales as the square of the bearing length and linearly as the bearing radial stiffness. As a result of combining the design study with the experimental data and existing bearing analyses, it has become possible to design a journal bearing and thrust plate for angular stiffness as well as radial and axial stiffness.

The definition of the required bearing parameters comes from a study of KT series system requirements and a basic understanding of aircraft navigation systems. Steady radial and axial load capacities of 30 g are arbitrarily considered acceptable. No aircraft can achieve steady accelerations of that sort. However, vibration loads can represent a significant fraction of 30 g.

Angular load capacity is defined by the system gimbal loop gain, and the built in test equipment, (BITE) that limits the angular velocity of runaway gimbals or gimbal erection activities. In the latest KT systems

the gimbal rate limit is 10 radians/sec. In earlier systems it was as high as 25 radians/sec. An important consideration from the viewpoint of a low cost retrofit of a liquid bearing gyro is that an existing gimbal velocity limiting system be used. Therefore the angular rate capability must be defined at a minimum of 10 radians per second.

Vibration resistance requirements are determined by the vehicle. A typical environment was defined by multiplying the KT-70 gimbal transmissibility by the A7 aircraft environment. Since the gimbal sets have slightly different behavior for different input directions, bands of frequencies are chosen that encompass resonances that are close to one another rather than showing a large number of peaks. This approach also has the advantage of accomodating system to system variations.

Gyro power consumption is specified for the KT-70 platform as 3.0 watt maximum running and 5.5 watts maximum for starting power. Existing ball bearing units consume roughly 2 watts.

Assuming the use of a reduced length 70 micro inch gap journal as described in Section 3, a power dissipation of 2.4 watts per gyro is expected. This produces an additional 0.8 watts on the platform cluster. Were this uncompensated it would produce a 3 deg F rise in the cluster temperature. However, by reducing the set point of the outer temperature controler by a like amount the same gradients will

be set up and the additional heat removed. Therefore the additional power dissipation of the stiffened liquid bearings can easily be accomodated.

The self vibration of the dry tuned rotor gyros in a platform must be controlled to avoid bias instability as described in Section 2. Ball bearing gyros produce noise over a wide band of frequencies including a significant component at twice wheel speed, the most significant frequency. The liquid bearing on the other hand has all its self vibration energy at wheel speed (see Section 4.3). Consequently, by maintaining the balancing specification for the liquid bearing gyro at the same value as the ball bearing gyro a significant improvement in performance is expected. In this light, the dynamic balance of the rotor is fixed at wheel speed at one dyne cm.

The following table summarizes the requirements on the gyro and bearing determined by physical compatibility considerations. The additional column headed "objective" indicates the values that are expected to be achieved, and in every case where a difference exists, that difference is in the direction of improved performance.

Parameter	Required Value	Objective
Journal length	.55 inch	.55 inch
Journal gap	.70 micro inch	.70 micro inch
Thrust plate diameter	.480	.480
Thrust plate gap	150 micro inch	150 micro inch
Axial load capacity	30 g	38
Radial load capacity	30 g	>100
Angular rate capability	10 rad/sec	10 rad/sec
Vibration Resistance	See Fig. 6.1	See Fig. 6.1
Power Consumption	3 watt	2.4 watt
Dynamic balance	1 dyne cm	1 dyne cm

7.0 SAFETY ASPECTS

The safety aspects of the liquid bearing gyro have been reviewed with the considerations of paragraph 5.8.2.1 of Mil-Std-882 in mind. The only possible sources of physical harm to bystanders as a result of a catastrophic failure are the kinetic energy in the rotor, the lubricant (less than 0.2 grams of Heptane) and the mercury in the seals (less than 0.3 grams).

The kinetic energy in the rotor is in the order of 15 joules. Since the rotor is confined within a sealed gyro it is extremely unlikely that it can break through the cover with such low energy.

The heptane is flammable. Its flash point is 25 degrees F. However, the very small quantity in the gyro makes this consideration trivial.

The mercury is not flammable. However, mercury is a toxic substance. In view of the small quantity involved and since it is confined in a sealed gyro, it doesn't present a hazard to bystanders.

☆ U. S. GOVERNMENT PRINTING OFFICE: 1976 — 757-001/20